Computational Fluid Flow analysis in Cryogenic Turbo expander

A Thesis Submitted in Partial Fulfilment of the
Requirements for the Award of the Degree of

Bachelor of Technology

in

Mechanical Engineering

by

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National Institute of Technology, Rourkela
Rourkela-769008, Odisha, India

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Under the guidance of

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CERTIFICATE

This is to certify that the project work entitled " Computational Fluid Flow analysis of Cryogenic Turbo Expander " by Ravikumar Senthooran has been carried out under my supervision in partial fulfillment of the requirements for the degree of Bachelor of Technology during session 2013-2014 in the Department of Mechanical Engineering, National Institute of Technology ,Rourkela and this work has not been submitted elsewhere for a degree.

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Place : Rourkela
Date :

Acknowledgement

I place on record and warmly acknowledge the continuous encouragement, invaluable supervision, timely suggestions and inspired guidance offered by my guide **Prof. Ranjit Kumar Sahoo,** Department of Mechanical Engineering, National Institute of Technology, Rourkela, in bringing this report to a successful completion. I consider myself fortunate to get a chance to work under the guidance of such a dynamic personality.

This project has enriched my life, giving me a chance to work under a new environment of ANSYS. This project gave me a great opportunity to build my fundamentals in Computational Fluid dynamics and Cryogenic Engineering.

I am really thankful to **Mr.Balaji Choudhury** and **Mr. Sachindra Rout** for their valuable suggestions and encouragement to carry out my project. They are very much patient to listen to my problems and provided suitable solutions. I am really grateful for their support throughout my project.

I would like to thank all my classmates, Faculty members of Mechanical Engineering Department for making my 4 years stay in NIT- Rourkela, wonderful and memorable.

Last but not the least, I thank my Parents and the almighty God whose blessing are always there with me throughout my life.

Date :

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Abstract

Cryogenic turbo expander is one of the components used in cryogenic plant to achieve low temperature refrigeration. Basic components in cryogenic Turbo expander are Turbine wheel, brake Compressor, shaft, nozzle, Thrust bearing, Journal Bearing etc. In expansion Turbine Temperature of gases decreases due to expansion and produces coldest level of Refrigeration.

This project is all about Computational Fluid flow analysis of high speed rotating turbine.
This involves with the three dimensional analysis of flow through a radial expansion turbine, using nitrogen as flowing fluid. Cfd packages, Bladegen, Turbogrid and CFX are used to carry out the analysis. Bladegen is used to create the model of turbine using available data of hub, shroud and blade profile. Turbogrid is used to mesh the model. CFX-Pre is used to define the physical parameters of the flow through the Turbo expander. CFX-Post is used for examining and analyzing results. Using these results variation of different thermodynamic properties like Temperature, Pressure, density, velocity etc inside the turbine can be seen.

Several graphs are plotted showing the variation of velocity, pressure, temperature, entropy and Mach number along streamline and span wise to analyze the flow through cryogenic turbine.

Contents

Nomenclature

D	Diameter - wheel	m
d	diameter - shaft	m
E	entropy	$J\ kg^{-1}\ K^{-1}$
h	enthalpy	Jkg^{-1}
K_e	free parameters	dimensionless
K_h	free parameters	dimensionless
LE	Leading Edge	
M	Mach number	dimensionless
N	rotational speed	rev/min
n_s	specific speed	dimensionless
P	pressure	Nm^{-2}
Q	volumetric flow rate	m^3s^{-1}
r	radius	m
S_E	energy source	$kg\ m^{-1}\ s^{-3}$
S_M	momentum source	$kg\ m^{-2}s^{-2}$
T	temperature	K
t	blade thickness	m
TE	Trailing Edge	
U	velocity magnitude	ms^{-1}
Z	number of vanes	dimensionless
ρ	density	kgm^{-3}
τ	shear stress	$kg\ m^{-1}\ s^{-2}$
ω	rotational speed	rad/s
θ	tangential coordinate	dimensionless

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Chapter 1

Introduction

1.1 Introduction of Turbo expander

To produce cryogenic refrigeration, turboexpander is used in all areas of gas and oil industries. It is a pressure reducing device that produces cryogenic temperature at the same time recovers energy from a plant stream in form of shaft power that can be used to drive other machinery such as compressor.

Though nature has provided an ample supply of gaseous raw materials in the atmosphere (oxygen, nitrogen) and beneath the earth's crust (natural gas, helium), we need to harness and store them for meaningful use. In fact, the volume of consumption of these basic materials is considered to be an index of technological advancement of a society. For large-scale storage, easy transportation and for low temperature applications liquefaction of the gases is essential. For making atmospheric gases like oxygen, nitrogen and argon in huge scale, low temperature distillation gives the most economical route from many points of view. Further, many industrially important physical processes from SQUID magnetometers and superconducting magnets to preservation of blood cells and treatment of cutting tools, require very low temperature[31]. The low temperature required for liquefaction of common gases can be acquired by several processes.

The low temperature can be achieved in many ways. During the first half of twentieth century helium and hydrogen liquefiers based on the high pressure Linde and Heylandt cycles were common in air separation plants. In recent years, cryogenic process plants are entirely based on the low pressure cycles. Expansion turbines are used to generate refrigeration. Turbine based plants have the benefits over high and medium pressure cycles. They are higher thermodynamic efficiency, higher reliability and easier integration with other system. Cryogenic plants may also use reciprocating expanders. The use of reciprocating expanders has been discontinued due to higher reliability and higher efficiency of small expansion turbines. Turbo expanders also provide refrigeration in various other applications, such as generating refrigeration to provide air conditioning to aircrafts. It is also used for seperation of propane and heavier hydrocarbon from natural gas streams. It generates low temperature required for the recovery of ethane in very less cost compared to any other method.

Expansion Turbines are widely used for energy extraction applications, Power cycles using geothermal heat, waste gas energy recovery in paper industries Organic Rankine cycle used in cryogenic process plants and freezing impurities in gas streams.

1.2 Anatomy of Cryogenic Turbo expander

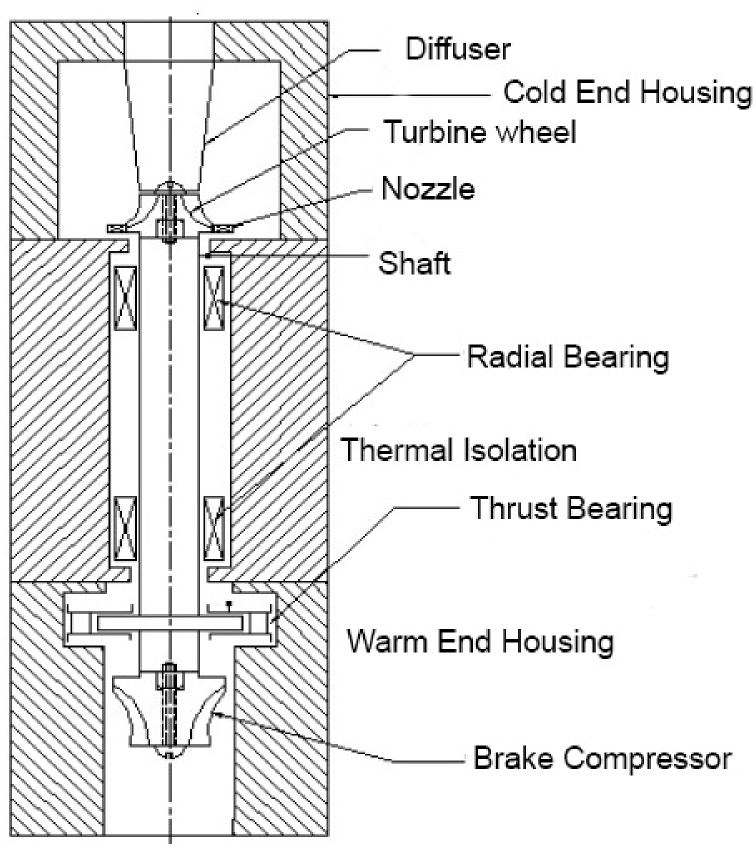


Fig 1.1 : Schematic of an expansion turbine

Turbo expander essentially contains turbine wheel and a brake compressor mounted on opposite sides of a single shaft and supported by required number of journal and thrust bearing. Using appropriate housing basic components are held in place and which also contains inlet and exit ducts.

The basic components are:

1.Turbine wheel	2.Diffuser	3.Nozzle
4.Compressor	5.Shaft	6.Journal Bearing
7.Thrust Bearing	8.Cold end bearing	9.Warm end Bearing
10.Bearing housing		

For easy installation and maintenance, most of the rotors for small and medium sized cryogenic plants are vertically oriented. The high-pressure gas enters the turbine through the channels, into the assembly of the cold end housing and from there it streams radially into the nozzle ring. The fluid accelerates through the converging passages of the nozzles. Static pressure is reduced due to transformation of pressure energy into kinetic energy. The high velocity fluid streams intrude on the rotor blades, imparting force to the rotor and creating torque. The nozzles and the rotor blades are arranged in such a way to eliminate sudden changes in flow direction and resulting loss of energy.

The turbine wheel considered in this investigation is of radial or mixed flow geometry, i.e. the flow enters the turbine wheel radially and exits axially. The blade passage has a profile of a three dimensional converging duct, changing from purely radial to an axial-tangential direction[31]. Work is extracted as the process gas under goes expansion with corresponding drop in static temperature.

Diffuser converts most of the kinetic energy of the gas leaving into potential energy in the form of pressure gain. Thus the pressure in the outlet of the system is lesser than the discharge pressure of the system. An electrical generator, eddy current brake, centrifugal compressor or an oil drum is used as loading device to extract work from turbine. In smaller systems energy is dissipated by connecting the discharge of compressor to the suction through a throttle valve and a heat exchanger. To avoid radial load in the bearings, Rotor is vertically mounted.

1.3 Aim of the Present work

Industrial gas manufacturers has substituted high pressure Linde and medium Pressure reciprocating engines based Claude system into low pressure cycle expansion turbines. Thus in modern days Expansion turbines are widely used in Cryogenic process plants.

The main motive of this investigation is developing a computational fluid flow analysis of turbo expander system. The objectives include: (1) Learning the Design and analysis software required for carrying out the investigation (2) Construction of the model (3) studying the performance of the model. A turbo expander system with following specifications has been taken for computational studies.

Working Fluid	Nitrogen (N_2)
Mass Flow Rate	0.024 kg/s
Turbine Inlet Temperature	99.65 K
Turbine Inlet Pressure	3 bar
Turbine Outlet Pressure	1.27 bar

Table 1: Specifications

1.4 Organization of the Thesis

The thesis is divided into six chapters. The Chapter 1 is brief introduction about the Cryogenic Turbo expander, their applications and aim of the work.

Chapter 2 describes the extensive survey of available literature on various aspects of cryogenic turbine development.

Chapter 3 describes the design procedure and graphical design of the turbine rotor. This chapter contains Design theory of the turbo expander, blade profile coordnates of turbo expander and Design of Turbo expander in Ansys Bladegen

Chapter 4 describes the Computational Fluid Flow analysis of Turbo expander. This includes meshing the model in Turbogrid and Simulation in CFX.

Chapter 5 describes the simulated results, variation of different thermodynamic properties through different graphs and contours.

Chapter 6 is limited to conclusions and the future works to be carried out.

Chapter 2

LITERATURE SURVEY

2. Literature Review

Expansion Turbines or Turbo expanders are the main components of cryogenic plants. It has attracted large number of researchers due to its extensive applications. Journals such as “Cryogenics and Turbo machinery” major conference proceedings like Advances in cryogenic engineering, Proceedings of International Cryogenic engineering contribute some of their portion for the research findings on Turbo expander technology.

2.1 History of Development

Liquefaction of gases was first introduced by Lord Reighleigh in his letter to “Nature” in1898. He discussed the use of Turbine instead of a piston expander. He highlighted that the most important function expansion turbine is producing refrigeration than power recovering. In1898, a liquefying machine was patented by, a British engineer, Eddar.C.Thrupp, using an expansion Turbine. It is a double flowing device cold air entering the center and dividing into two oppositely flowing stream. Joseph E. Johnson in USA patented an apparatus for liquefying gases. Joseph’s expander was a De Laval or single stage impulse turbine. In 1934, a report was published on successful commercial use of cryogenic expansion turbine at the Linde works in Germany [1]. In a low pressure air liquefaction and separation cycle, single stage axial flow turbine was used. An inward radial flow impulse turbine replaced it after two years.

In 1939 Kaptiza published description of a low temperature turbo expander. In USA in 1942 National Defence Research Committee sponsored for developing a turbo expander which operated without trouble for periods aggregating 2,500 hrs and achieved an efficiency of more than 80%. In 1958 radial inward flow turbine was developed by the United Kingdom Atomic

Energy Authority for a nitrogen production plant [2]. In 1964, the first commercial turbine using helium was started operating in a refrigerator. 73 W at 3K was produced by this turbine for the helium bubble chamber. A turbine was developed by National Bureau of Standards at Boulder, Colorado [3] with shaft diameter of 8 mm. This turbine was operated at a speed of 600,000 rpm at 30 K inlet temperature. In 1974, a turbo expander with self-acting gas bearings was developed by Sulzar brothers in Switzerland [4]. In 1984, experimental testing was carried out in the prototype turbo expander of medium size in a nitrogen liquefier. A micro turbo expander for a small helium refrigerator was constructed for a small helium refrigerator based on Claude cycle by Izumi et. al [5] in Japan. This turbo expander comprised of a radial inward flow reaction turbine and centrifugal brake fan fixed on the lower and upper ends of a shaft. This centrifugal fan was supported by self-acting gas bearings. In 1979, Kun & Sentz [6] initiated studies to survey operating plants and cost factors were generated. Miniature turbines for Brayton Cycle crycoolers were developed by Sixsmith et. al[7] in association with Goddard Space Centre of NASA.

Kate et. al [8] developed the turbo expander with variable flow capacity mechanism (an adjustable turbine). the Naka Fusion Research Centre affiliated to the Japan Atomic Energy Institute [9-10] developed a wet type helium turbo expander with projected adiabatic efficiency of 70%. In 1991, third stage tubo expander was designed and manufactured for the gas expansion machine regime by “Cryogenmash” [11]. Design of each stage of turbo expander was almost similar, they differ each other by dimensions produced by “Heliummash” [11]. Agahi et. al. [12-13] have discussed the design process of the turbo expander using modern technology, such as

Computational Fluid Dynamic software, Computer Numerical Control Technology and Holographic Techniques.

A small wet turbine was developed by Sixsmith et. al. [14] at Creare Inc., USA for a helium Liquefier set up at particle accelerator of Fermi National laboratory. The expander shaft was supported in pressurized gas bearings and had a turbine rotor of 12.7 mm brake compressor at the warm end and 4.76 mm at the cold end. The design speed and design cooling capacity of turbo expander are 384,000 rpm and 444 Watts respectively. A cryogenic turbo expander with a 103 mm long rotor and weight of 0.9 N was developed by Xiong et. al.[15] at the institute of cryogenic Engineering, China. It had a working speed up to 230,000 rpm. The turbo expander was tested with two types of gas lubricated foil journal bearings.

India has been lagging behind the rest of the world in this field of research and development. Still, during the past two decades significant and decent development has been progressed. In CMERI Durgapur, Jadeja et. al [16-17] established an inward flow radial expansion turbine supported on gas bearings for cryogenic plants. This device resulted stable rotation at 40,000 rpm. PhD dissertation of Ghosh [18] explains the detailed summary of technical features developed in various laboratories. Helium refrigerator was established recently by Cryogenic Technology Division, BARC, which is capable of producing 1 kW at 20K temperature.

2.2 Design and Development

Design of Turbo expander is generally based on several engineering disciplines like fluid dynamics, mechanical vibration, tribology, stress analysis, controls, mechanical design and fabrication. Design parameters that contribute designing are mass flow rate, gas composition, inlet- outlet pressure, inlet temperature.

During the past two decades, performance chart has become commonly accepted mode of presenting characteristics of turbo machines [20]. Several characteristics values are used for describing significant performance criteria of turbo machines such as turbine velocity ratio, pressure ratio, flow coefficient and specific speed [20]. For computing the efficiency of radial turbo machines a simplified method was presented by Balje to calculate their characteristics [21]. Similarity principles portrays that two parameters are adequate to determine major dimensions as well as the inlet and exit velocity triangles of the turbine wheel.

The specific speed and the specific diameter completely define dynamic similarity [31]. Specific Speed parameter was introduced by Balje[22] in design of gas turbines and compressors. Values of specific speed and specific diameter may be selected for getting the highest possible polytropic efficiency and to complete the optimum geometry [19] The ratio of exit tip to rotor inlet diameter should be limited to a maximum value of 0.7 to avoid excessive shroud curvature. Similarly, the exit hub to the tip diameter ratio should have a minimum value of 0.4 to avoid excessive hub blade blockage and loss [43, 45]. Kun and Sentz [6] have taken $\varepsilon = 0.68$. Balje [7] has taken the ratio of exit meridian diameter to inlet diameter of a radial impeller as 0.62.Balje has derived an equation for the minimum rotor blade number as a function of specific speed. Denton [25] has given guidance on the choice of number of blades. He suggests that a number of 12 blades is usual for cryogenic turbine wheels. Twelve complete blades and twelve partial blades were used by Sixsmith [3] in his turbine designed for medium size helium liquefiers. The blade number is calculated from the value of slip factor [16]. The number of blades must be adjusted so that the blade width and thickness can be manufactured with the available machine tools.

Chapter 3

Design Theory

&

Graphical Design

Design Theory

3.1 Turbo expander design

Some factors play an important role in the design procedure of a turbo expander. They are type of working fluid, rate of fluid flow, Inlet, outlet conditions and expansion ratio. Design procedure in this chapter allows any arbitrary value of combination of fluid species, inlet conditions as they are adequately taken care in the equations. Design methodology contains following sections. They are (1) Fluid parameters & layout of the components (2) Turbine wheel design and determination of blade profile.

Fluid parameters and layout of components

Fluid parameters considered in this chapter are suitable for a small refrigeration unit that produces less than 1kW refrigeration. The inlet temperature has been selected arbitrarily in such a way that even with ideal expansion the exit state should not fall in two phase region. Basic parameters for the cryogenic turbo expander are given in the table below.

Working Fluid	Nitrogen (N_2)
Mass Flow Rate	0.024 kg/s
Turbine Inlet Temperature	99.65 K
Turbine Inlet Pressure	3 bar
Turbine Outlet Pressure	1.27 bar

Table 3.1.1 : Basic parameters

Turbine wheel design and Determination of Blade profile

Balje [8] and Kun & Sentz [29] stated a method for turbine wheel design, which are based on the similarity principles. Similarity principle states that to achieve an optimized geometry for a maximum efficiency, for a given Reynolds number, Mach number and specific heat ratio of the fluid, two dimensionless parameters: specific speed and specific diameter helps to determine the major dimensions of the wheel and its inlet and exit velocity triangles[31]. Specific speed and specific diameter are defined below.

Specific Speed $$n_s = \frac{\omega \times \sqrt{Q_3}}{(\Delta h_{in-3s})^{3/4}}$$

Specific Diameter $$d_s = \frac{D_2 \times (\Delta h_{in-3s})^{1/4}}{\sqrt{Q_3}}$$

Balje [8] has stated that for the maximum efficiency of radial inflow turbine the values of specifc speed and specific diameter are

$$n_s = 0.54 \ \& \ d_s = 3.4$$

Major Dimensions calculated for the prototype turbine by Ghosh [52] are given below.

Rotational Speed: 218790 rpm

Wheel Diameter $D_2 = 16\ mm$

To avoid excessive shroud curvature, the ratio of exit tip diameter to inlet tip diameter should have a maximum value of 0.7 [25]. Corresponding to the peak efficiency point (Balje,1981):

$$\frac{D_{tip}}{D_2} = 0.676$$

$$D_{tip} = 10.8 \text{ mm}$$

In order to determine the optimum turbine geometry, various studies have been conducted and published in relation to the rotor blade profile (Thakker and Abdulhadi, 2007). Rohlik and Harold (1968) stated that the exit hub to tip diameter ratio should maintain a minimum value 0.4 to avoid excessive hub blade blockage and energy loss.

$$\frac{D_{Hub}}{D_{Tip}} = 0.425$$

$$\mathrm{D}_{Hub} = 4.6\,mm$$

Number of blades: 10

Blade Thickness : 0.6 mm

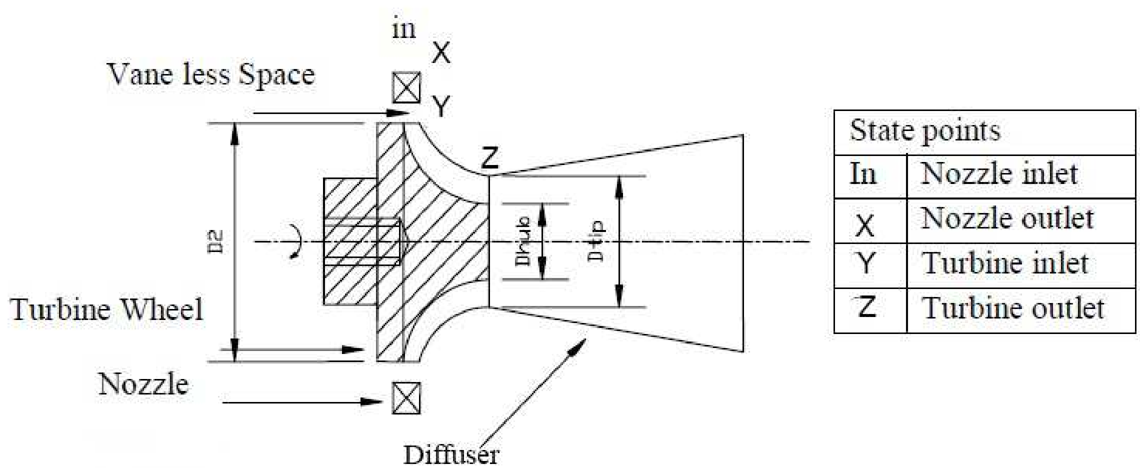


Fig 3.1.1 : State points of turbo expander

Computation of 3D contours of blades is described below. The computational procedure suggested by Hasselgruber [27] and extended by Kun & Sentz [6] has been adopted. Length and curvature of the flow path causes the fluid pressure loss in the turbine blade passage. Parameters defined by Hasselgruber [27] K_e and K_h control the flow path and curvature. The optimum of the blade profile is determined by magnitude of the velocity and change in its direction. For the turbine blade design K_e varies between 0.75 and1 ,K_h varies between 1 and 20. S. K. Ghosh [29] states that an ideal parameters Ke= 0.75 and Kh= 5.0 provides a better profile for turbo expander. The blade profile co-ordinate of pressure surface, mean surface and suction surface are shown in below tables[29]

TIP CAMBER LINE			HUB CAMBER LINE		
Z(MM)	R (mm)	Q (Deg)	Z(mm)	R(mm)	Q(Deg)
-0.24	5.38	0	0.24	2.32	0
0.24	5.29	6.71	0.67	2.56	6.71
0.71	5.22	12.39	1.11	2.76	12.39
1.18	5.19	17.19	1.55	2.94	17.19
1.63	5.18	21.22	2	3.1	21.22
2.08	5.19	24.58	2.46	3.25	24.58
2.52	5.22	27.37	2.93	3.4	27.37
2.95	5.27	29.65	3.39	3.56	29.65
3.37	5.33	31.49	3.86	3.72	31.49
3.79	5.41	32.96	4.33	3.91	32.96
4.19	5.51	34.1	4.79	4.13	34.1
4.58	5.63	34.96	5.24	4.37	34.96
4.97	5.78	35.61	5.68	4.65	35.61
5.34	5.95	36.06	6.09	4.97	36.06
5.69	6.16	36.38	6.47	5.32	36.38
6.02	6.9	36.58	6.81	5.7	36.58
6.33	6.68	36.7	7.11	6.11	36.7
6.62	6.99	36.77	7.37	6.54	36.77
6.87	7.33	36.8	7.59	6.99	36.8
7.09	7.7	36.81	7.79	7.45	36.81
7.28	8.1	36.81	7.9	7.92	36.81

Table 3.1.2: Coordinates of blade profile

Z pressure(mm)	R pressure (mm)	Q Pressure (rad)	Z suction (mm)	R suction (mm)	Q suction
0	3.85	0.055	0	3.85	-0.055
0.45	3.91	0.339	0.45	3.92	0.068
0.91	3.99	0.404	0.91	3.99	0.172
1.36	4.67	0.458	1.36	4.07	0.261
1.82	4.14	0.502	1.82	4.14	0.336
2.27	4.22	0.537	2.27	4.22	0.4
2.72	4.31	0.566	2.72	4.31	0.453
3.17	4.41	0.588	3.17	4.41	0.497
3.62	4.53	0.665	3.62	4.53	0.533

4.06	4.66	0.617	4.06	4.66	0.562
4.49	4.82	0.627	4.49	4.82	0.585
4.91	5	0.633	4.91	5	0.603
5.32	5.21	0.637	5.32	5.21	0.616
5.71	5.46	0.64	5.71	5.46	0.626
6.08	5.74	0.61	6.08	5.74	0.633
6.42	6.05	0.642	6.42	6.05	0.637
6.72	6.39	0.642	6.72	6.39	0.64
6.99	6.77	0.642	6.99	6.77	0.641
7.23	7.16	0.642	7.23	7.16	0.642
7.43	7.58	0.642	7.43	7.581	0.642
7.59	8.01	0.642	7.59	8.01	0.642

Table 3.1.3 : Coordinates of Pressure and suction surface

3.2 Design of turbine wheel in Bladegen

BladeGen is a component of ANSYS BladeModeler. It is a geometry creating tool. Using Hub, shroud and blade profile coordinates mentioned in previous tables the model was created in bladegen. Joining the hub and shroud streamlines the surface is created.

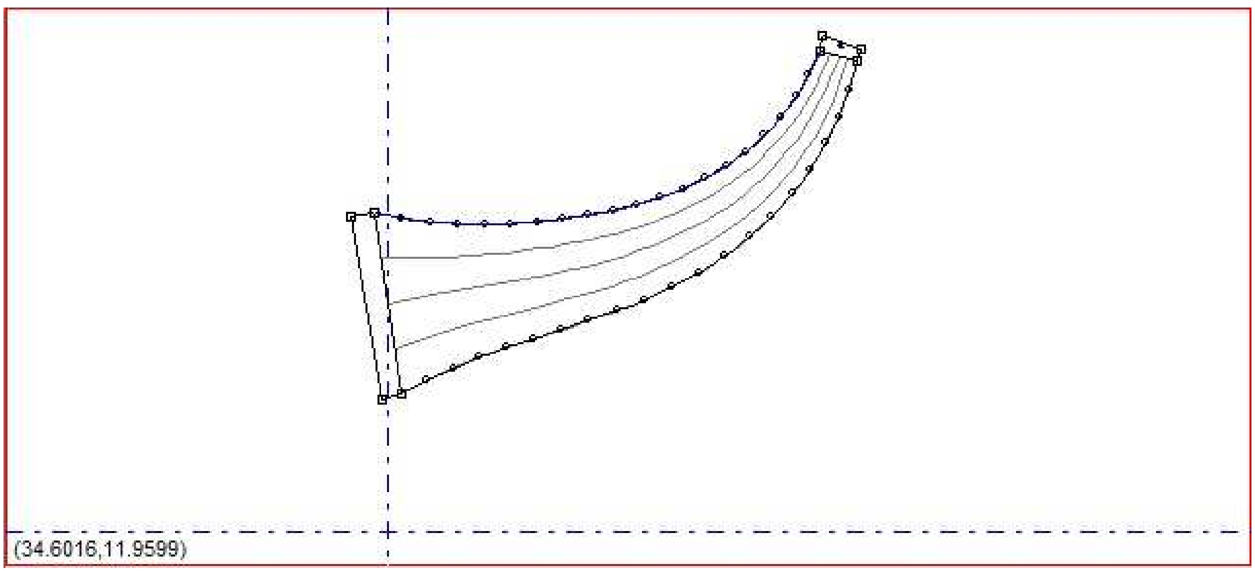


Fig 3.2.1 Meridional View of the blade profile

The meridional view contains the description of the blade in an axial-radial coordinate system.

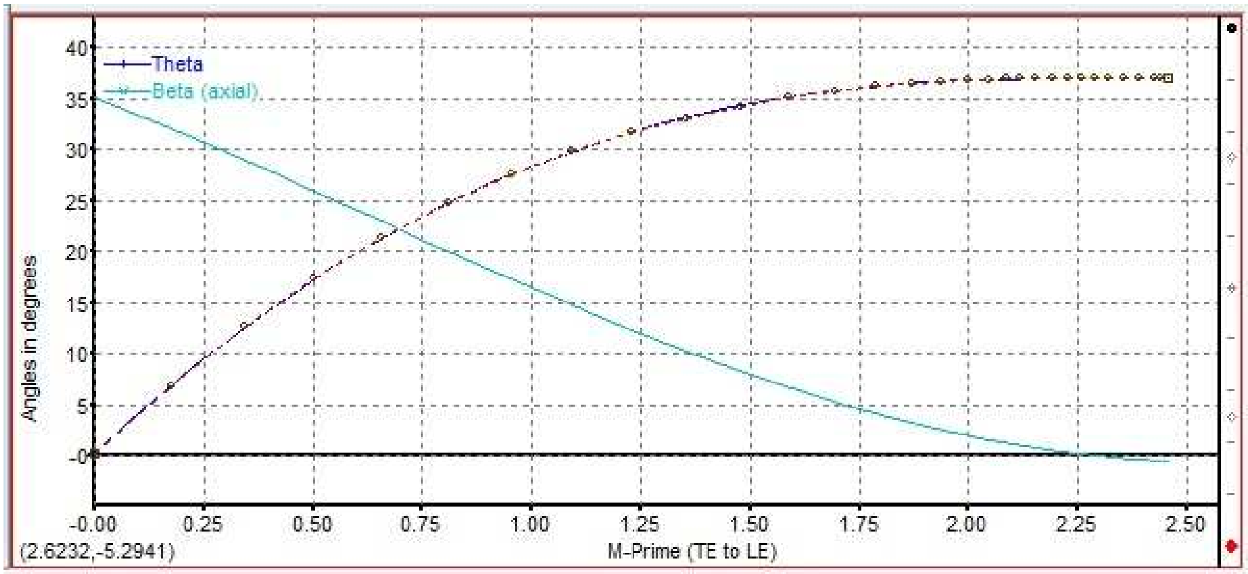


Fig 3.2.2 Hub blade angle variation in Angle view of Bladegen

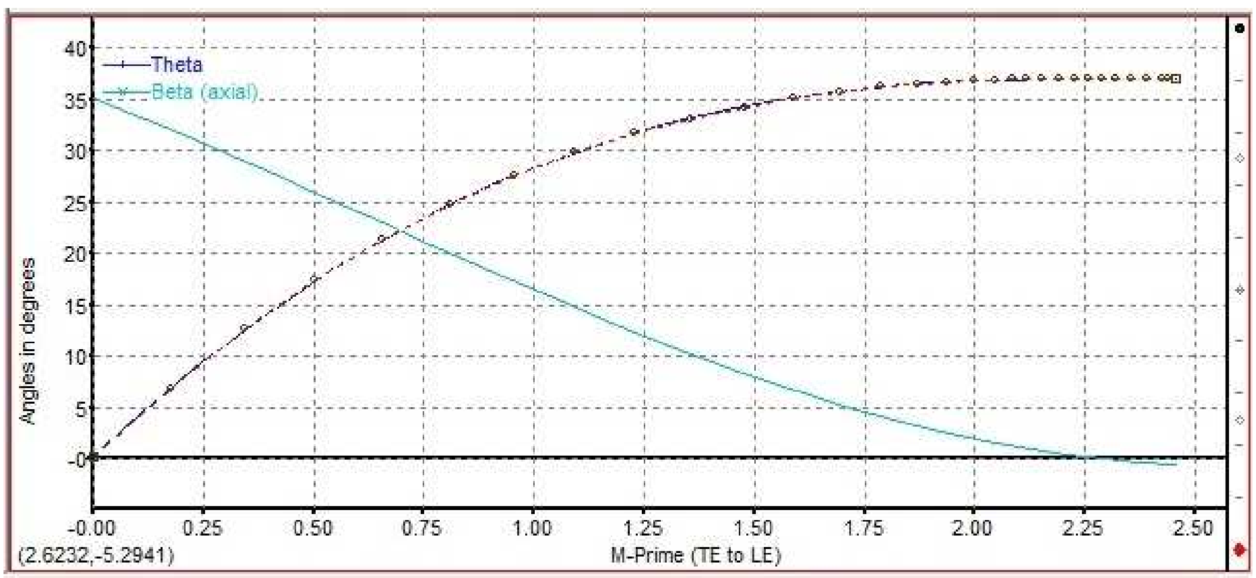


Fig 3.2.3 Shroud blade angle variation in Angle view of Bladegen

Chapter 4

Computational Fluid flow Analysis

Computational Fluid flow analysis is done in three steps. First step is the designing part in **Bladegen** which has been discussed in the previous chapter. Next is meshing the model using **Turbogrid**. Then **CFX-Pre** is used to define the simulation settings and physical parameters required for the flow through the turbo expander. CFX-Post helps to analyze the results.

4.1 Meshing the model

Turbogrid is used to mesh the model. High quality hexahedral meshes are created which fulfills the demands of fluid flow analysis in turbine rotor. Turbine rotor geometry is imported from bladegen. Basic information about the geometry is given in machine data. Here unit has been selected in mm. These units are used for the internal representation of the geometry to reduce the computer round-off errors. In the details of the Shroud tip, tip option has been selected as Constant span and span = 0.985 to create a gap between blade and shroud tip.

Next is creating the topology. Topology created guides the mesh. In Topology definition placement has been selected as " Traditional with control points". This option provides access to the legacy topology method. Then method was selected as "H/ J/C/ L Grid method". The H/J/C/L-Grid method causes ANSYS TurboGrid to choose an H-Grid, J-Grid, C-Grid, L-Grid, or a combination of these, based on heuristics. It chooses a J-Grid topology for the upstream end of the passage, and H-Grid topology for the downstream end. Selecting Include O-grid ensures the O-Grid around the blade to increase mesh orthogonality in that region (Ansys help). O-Grid thickness is made equal to half the average blade thickness betting width factor equal to 0.5.

Periodicity > Projection was set to Float ton Surface. This allows the periodic surface of the mesh to deviate from the geometric periodic surface, in order to improve mesh skewness properties along the periodic boundary.

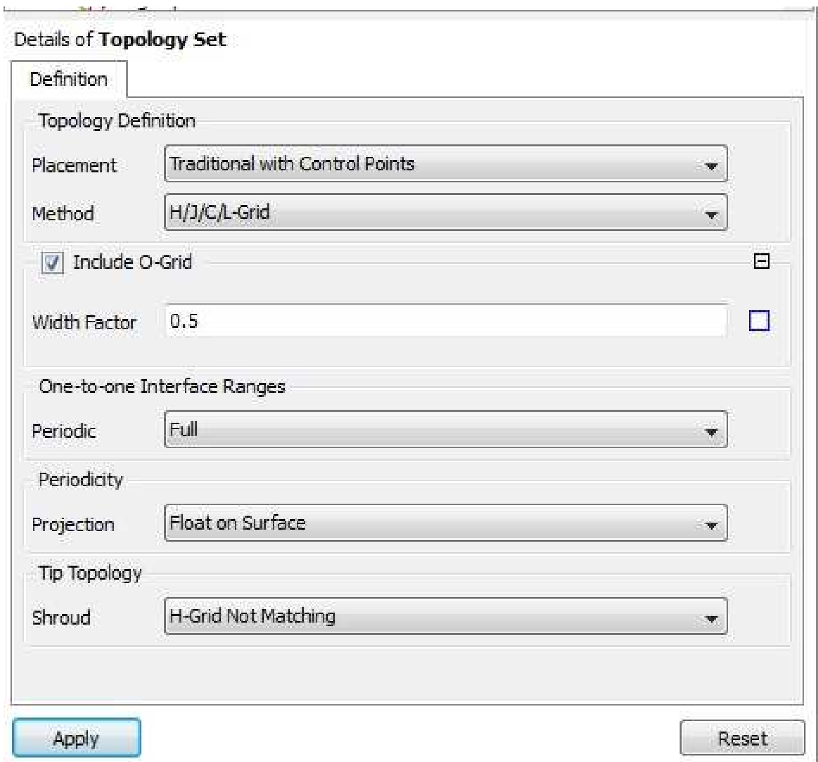


Fig 4.1.1: Topology setting

After setting the topology, the target number of mesh was set to 250000 to get a fine mesh. Before generating the 3D mesh, the quality of the mesh in hub and shroud layers should be checked. After correcting the mesh quality we generate the mesh with 266070 nodes and 242136 elements.

4.2 Defining Physical parameters using Ansys CFX-Pre

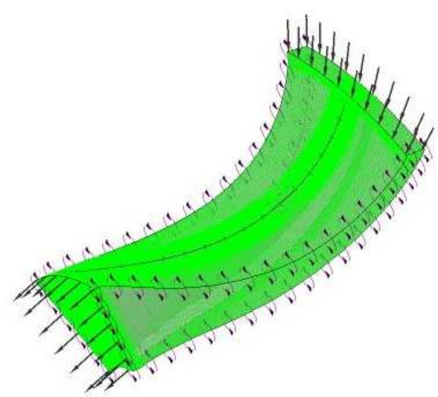
In CFX-Pre Turbo mode is selected to define physic of meshed Turbine Rotor. In the basic settings of Turbo mode machine type is set as Radial Turbine and rotation axis is set as z- axis. In component setting component type is set as rotating and value of rotation is 218780 rev / min. Turbo mode automatically selects regions and identical boundary types. Fluid type, analysis type, model data, inflow and outflow boundary templates and solver parameters are set in physics definition tab.

Setting		**Values**
Fluid		Nitrogen
Analysis Type		Steady state
Reference Pressure		0 Pa
Heat transfer		Total energy
Turbulence		K-Epsilon
Inflow/ Outflow boundary templates		Mass flow inlet P-static outlet
Inflow	T-Total	99.65 K
	Mass flow	Per component
	Mass flow rate	0.0024 kg/sec
	Flow direction	Normal to boundry
Out flow P static		1.27 bar
Advection Scheme		High Resolution
Convergence control		Physical timescale
Physical Timescale		0.000004s

Table 4.2.1 Physics Definition

After setting the physics definition CFX-Pre generates appropriate interfaces and boundary conditions using region names

Fig 4.2.1 Flow direction



4.3 CFX Solver & CFX- Post

CFX-Solver is used to launch solvers and also monitors the output. ANSYS Workbench generates the CFX-Solver input file and passes it to ANSYS CFX-Solver Manager. In CFX solution units are set as SI system. Following are selected in Solver control dialog box :

Advection scheme: high resolution

Max iterations: 10000

Physical timescale: 4e-006[s]

Convergence criteria: Residual type : RMS

Residual target: 0.0001

In Output control : Extra output variable: Temperature

CFX –post is used to allow easy visualization and quantitative analysis of results of CFD simulations. Turbo workspace is used to improve and speed up post-processing for turbo machinery simulation. It includes plans, isosurfaces, vectors, streamlines, contours, animations, etc. It allows precise quantitative analysis as, weighted average, forces, results, comparisons, built in anduser defined macros. CFD-Post includes automatic reports, charts, and tables. In CFD-Post general workflow locations are prepared to extract data and to generate plots. Data is extracted by creating variable and expression at particular location. Qualitative and quantitative data is generated at that location and on the basis of it reports are generated.

Chapter 5

Results and Discussion

Results and Discussion

CFX – post is used to do the Computational Fluid Flow analysis after completion of CFX-solver. A tabulated result is generated in CFX- post. Different properties along inlet to Outlet and Hub to Shroud , graphs and Contours are generated in the report. Variation of different properties at different locations of Turbine rotor are stated below.

Quantity	Inlet	LE Cut	TE Cut	Outlet	TE/LE	TE-LE	Units
Density	8.8914	8.5773	4.8030	4.5589	0.5600	-3.7743	$\mathrm{kg}m^{-3}$
P Static	254695	246284	128044	126719	0.5199	-118240	Pa
P Total	299140	296209	174089	170029	0.5877	-122120	Pa
P Total (rot)	298907	292968	190798	186117	0.6513	-102170	Pa
T Static	91.0744	90.6845	89.4312	90.0614	0.9862	-1.2534	K
T Total	99.6537	100.0280	97.3913	97.4597	0.9736	-2.6369	K
T Total (rot)	99.6288	99.6346	99.6425	99.7145	1.0001	0.0079	K
H static	-214902	-215307	-216608	-215954	1.0060	-1301.06	Jkg^{-1}
H total	-205996	-205608	-208345	-208274	1.0133	-2737.17	Jkg^{-1}
U	189.4200	183.1320	94.5749	93.1323	0.5164	-88.5572	$\mathrm{m}s^{-1}$
Entropy	5378.37	5385.03	5517.3	5525.2	1.0246	132.2710	$\mathrm{Jkg}^{-1}\mathrm{K}^{-1}$
Mach(abs)	0.6679	0.7092	0.6561	0.6284	0.9252	-0.0531	
Mach (rel)	1.1895	1.1729	0.8903	0.8692	0.7591	-0.2825	
Cm	101.6300	114.8540	117.2040	82.8654	1.0205	2.3500	$\mathrm{m}s^{-1}$

Cu	14.8409	22.3988	-18.7815	1.7835	-0.8385	-41.1803	ms^{-1}
C	111.7830	130.4420	121.3810	98.2590	0.9305	-9.0607	ms^{-1}
Distortion Parameter	1.4100	1.2189	1.1290	1.3982	0.9262	-0.0899	
Flow Angle: Alpha	18.5584	17.7267	-8.6439	26.4361	-0.4876	-26.3706	[degree]
Flow Angle: Beta	-5.0897	-41.6126	-44.8204	12.4967	1.0771	-3.2078	[degree]

Table 5.1 Variation of Thermodynamic properties

Static Temperature of Nitrogen at outlet is 90.0614 K. which is almost near to the temperature obtained by S.K.Ghosh during his experimental work. Various graphs are obtained from results generated in CFX.

5.1.1 Variation of Pressure

Along Stream wise Inlet to Outlet

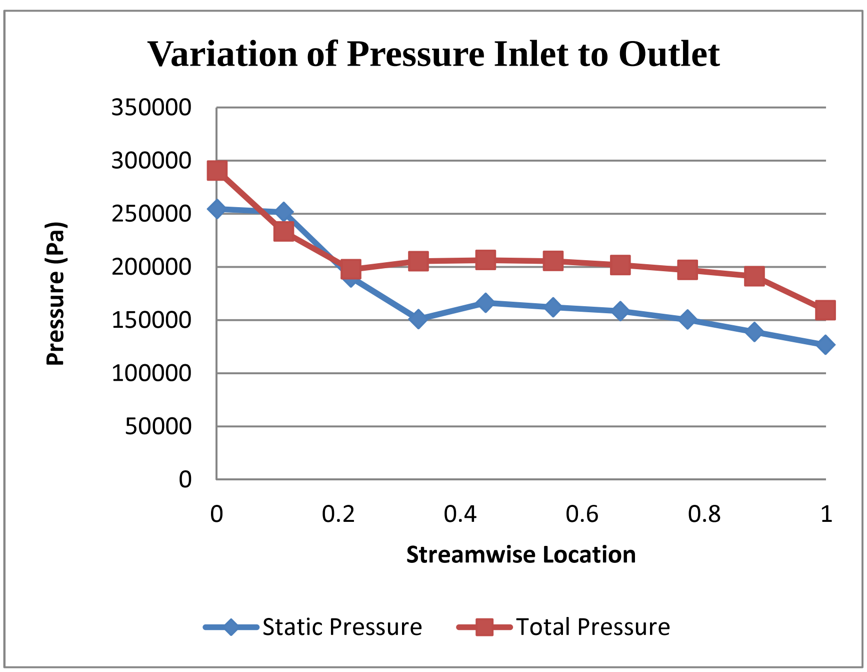


Fig 5.1.1 Variation of Pressure along stream wise inlet to outlet

Variation of Static pressure and Total pressure are shown in the graph above. Total pressure in the Inlet and Static pressure in the outlet are almost similar to the experimental results obtained by Ghosh [52]. Variation of Total Pressure is from 3 bar to 1.7 bar and Variation of static pressure is from 2.54 bar to1.27 bar.

Along Span wise Hub to Shroud

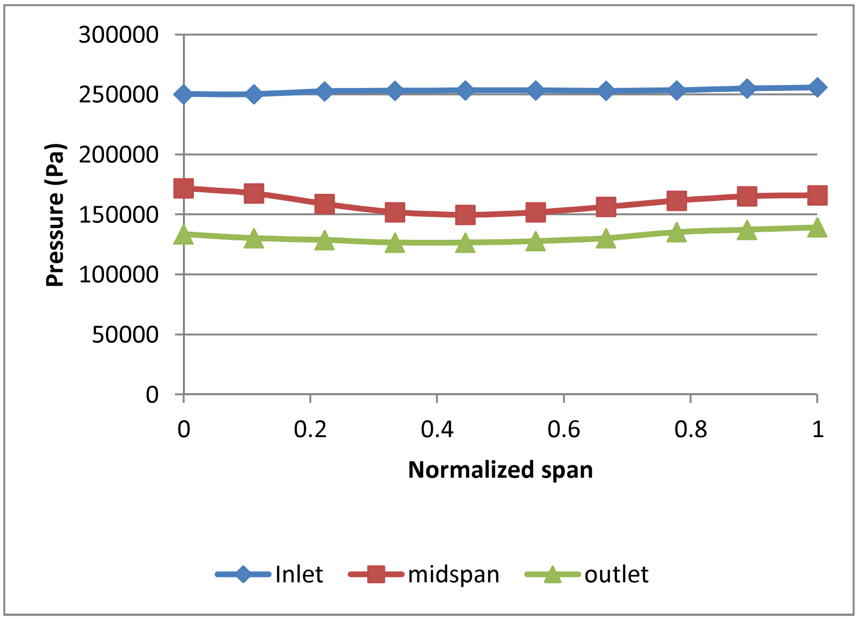


Fig 5.1.2 Variation of Pressure along span wise Hub to Shroud

Variation of Pressure along span wise Hub to shroud is shown in the above graph.

Inlet 2.5 to 2.6 bar , Midspan 1.65 to1.71 bar and Outlet 1.26 to 1.3 bar. are pressure variations obtained.

5.1.2. Variation of Temperature

Along stream wise Inlet to Outlet

Total temperature varies between 99.65K and 97.45K. Total static Pressure varies from 95 K to 88.9 K.

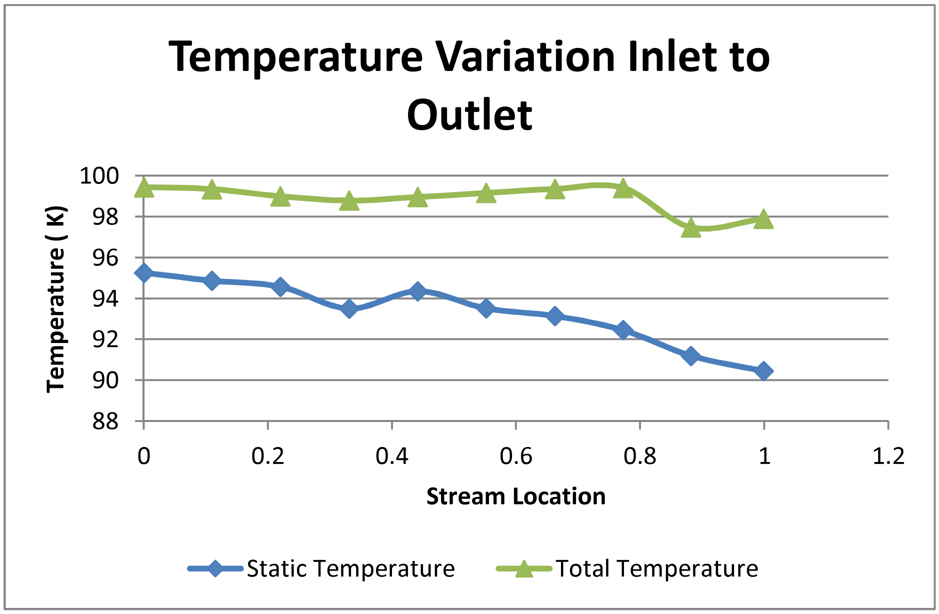


Fig 5.1.3 Variation of Temperature along streamwise Inlet to Outlet

Along span wise Inlet to Outlet

At inlet span hub to shroud variation temperature variation is 102.5 K to 101K, At mid span temperature variation 94 K to 96 K and outlet 92 K to 91 K.

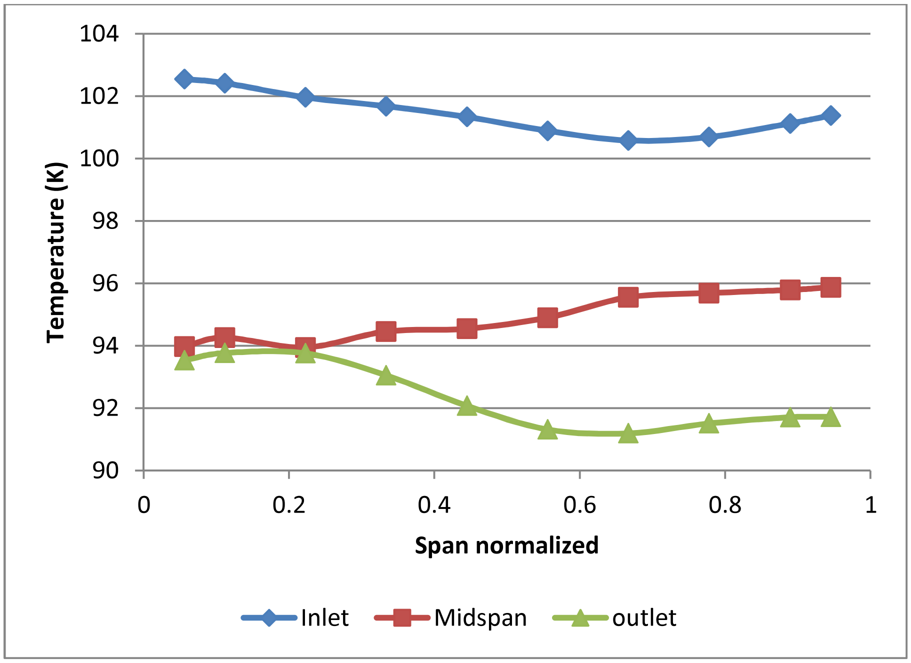


Fig 5.1.4 Variation of Temperature along spanwise Hub to Shroud

Velocity Variation

Along Stream wise Inlet to Outlet

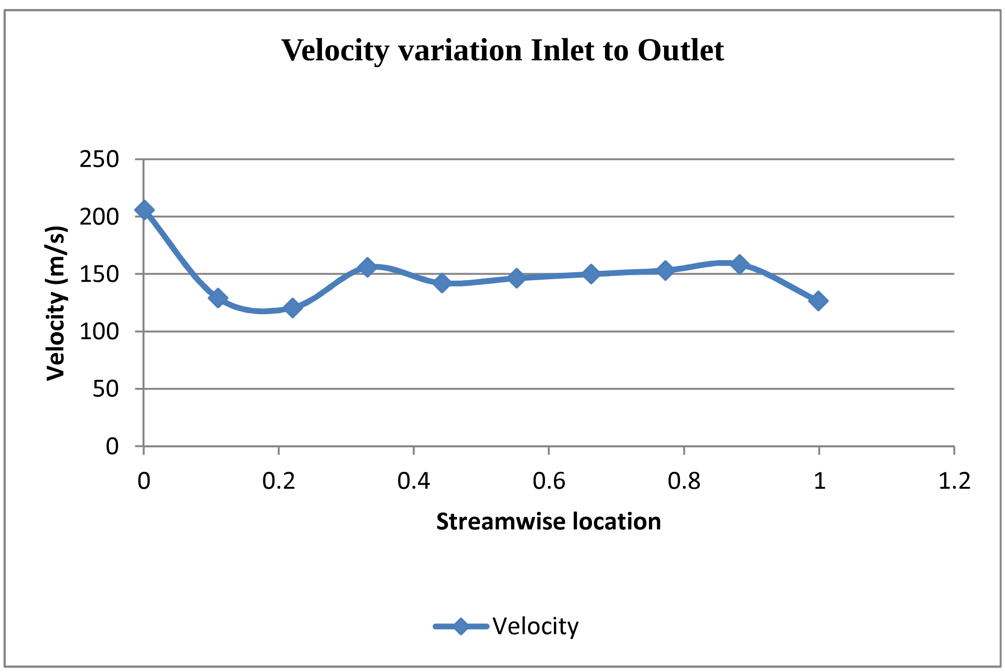


Fig 5.1.4 Variation of Velocity along Streamwise Inlet to Outlet

The above graphs show the velocity variation along Streamwise inlet to outlet.

Velocity is decreasing from 189.42 ms^{-1} to 133.26 ms^{-1} .

Along Span wise Hub to Shroud

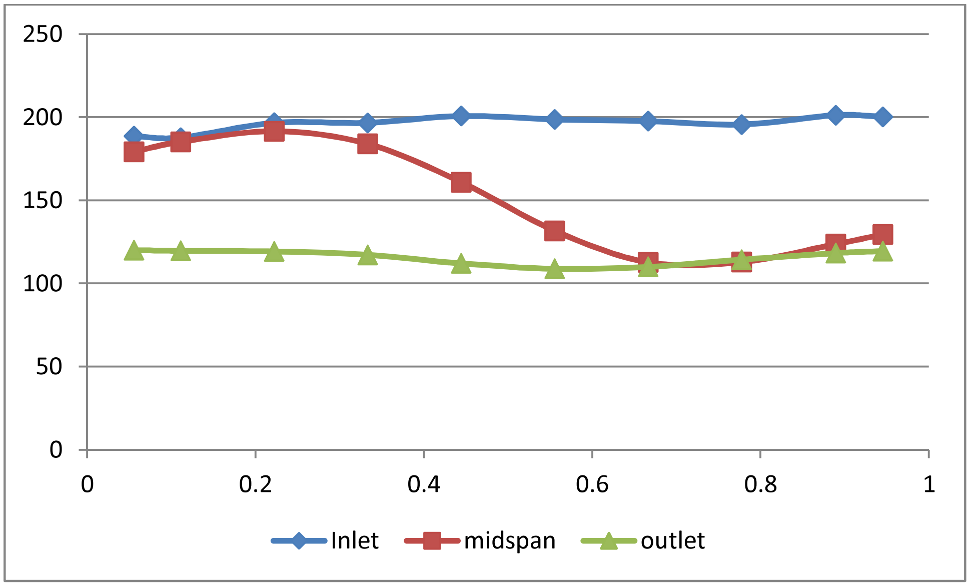


Fig 5.1.5 Variation of Velocity along Span wise Hub to Shroud

In inlet span the velocity varies from 190 ms^{-1} to 200 ms^{-1}, mid span velocity varies from 130 ms^{-1} to 150 ms^{-1} and in oulet span velocity varies from 100 ms^{-1} to110 ms^{-1}

Density Variation

Along Stream wise Inlet to Outlet

Density variation along stream wise Inlet to Outlet is shown in the graph above. Density is decreasing Inlet to Outlet, 8.89 kgm^{-3} to 4.56 kgm^{-3} respectively

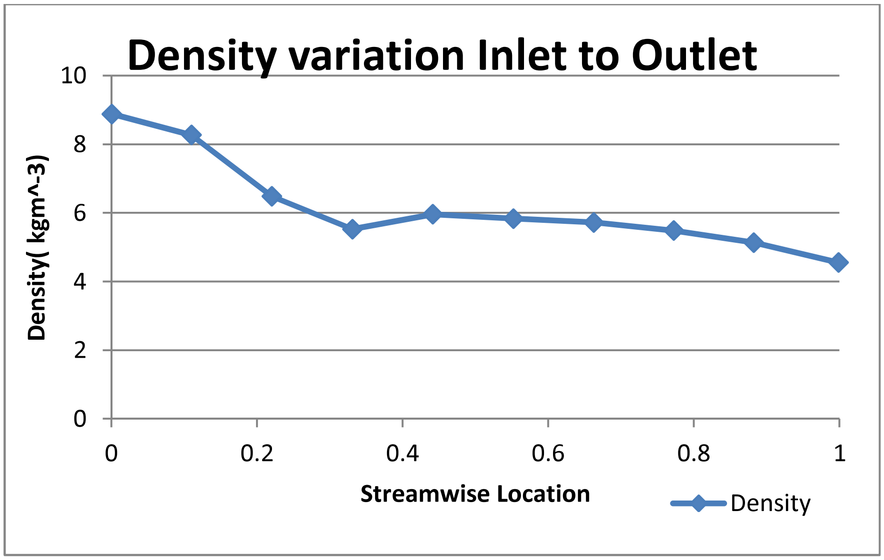


Fig 5.1.6 Variation Density along streamwise

Mach no Variation

Along Stream wise Inlet to Outlet

Variation of absolute and relative mach no is shown in the graph above. Absolute mach no varies from 0.66 to 0.62 and Relative mach no from 1.18 to 0.75 inside the rotor along stream wise inlet to outlet.

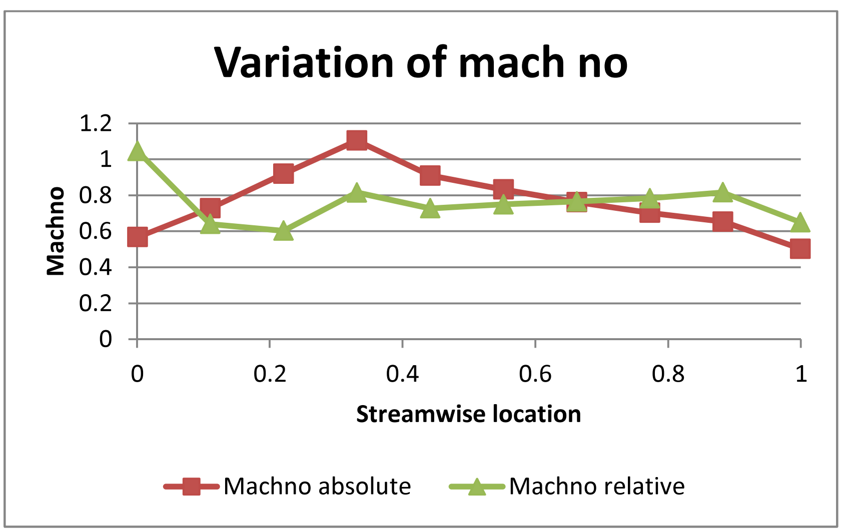


Fig 5.1.7 Variation of mach no along streamwise

Varaiation of Entropy Inlet to Oulet

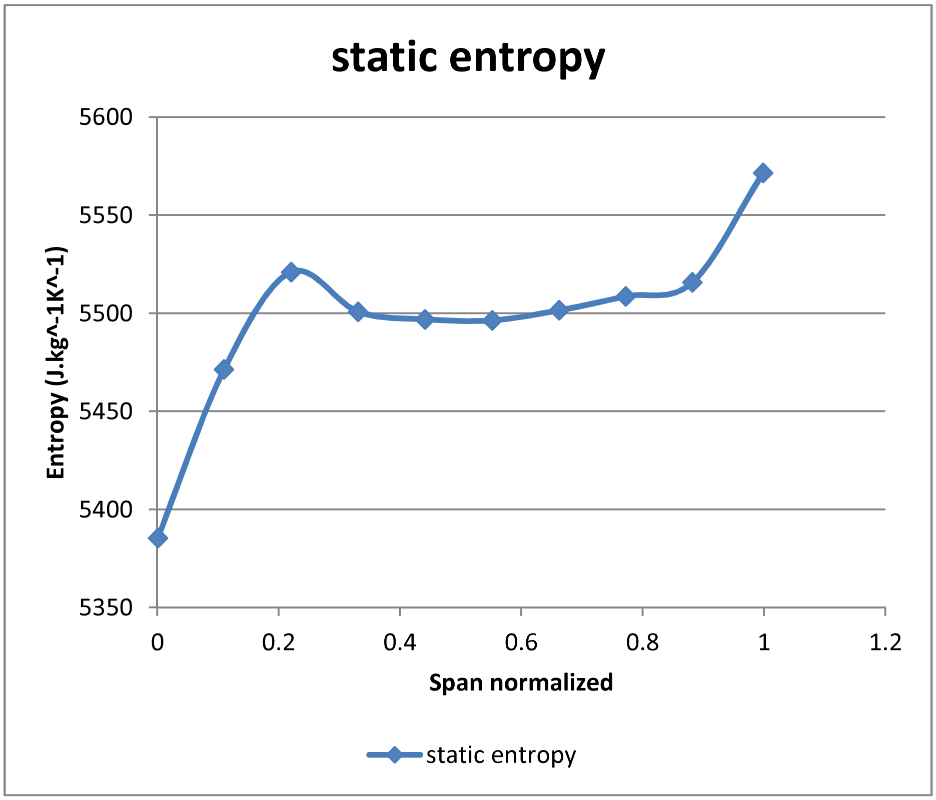


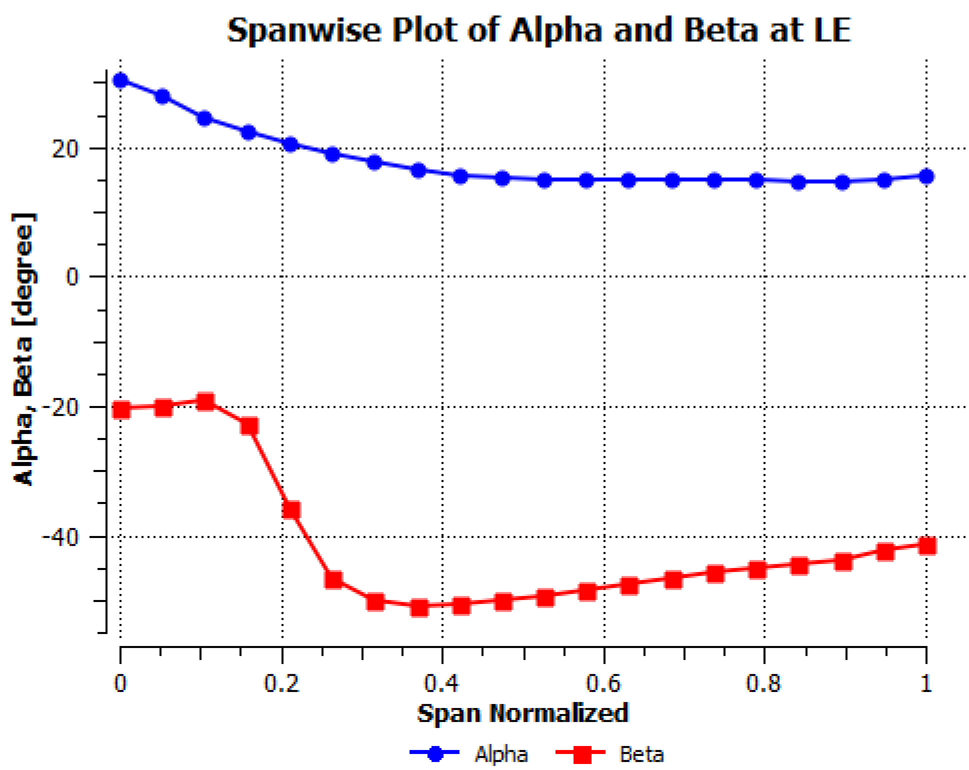
Fig 5.1.8 Variation of Entropy along stream wise

The above graph shows the variation of static entropy along stream wise inlet to outlet. Static entropy varies from 5378 kj/kg.K to 5575 kj/kg.K

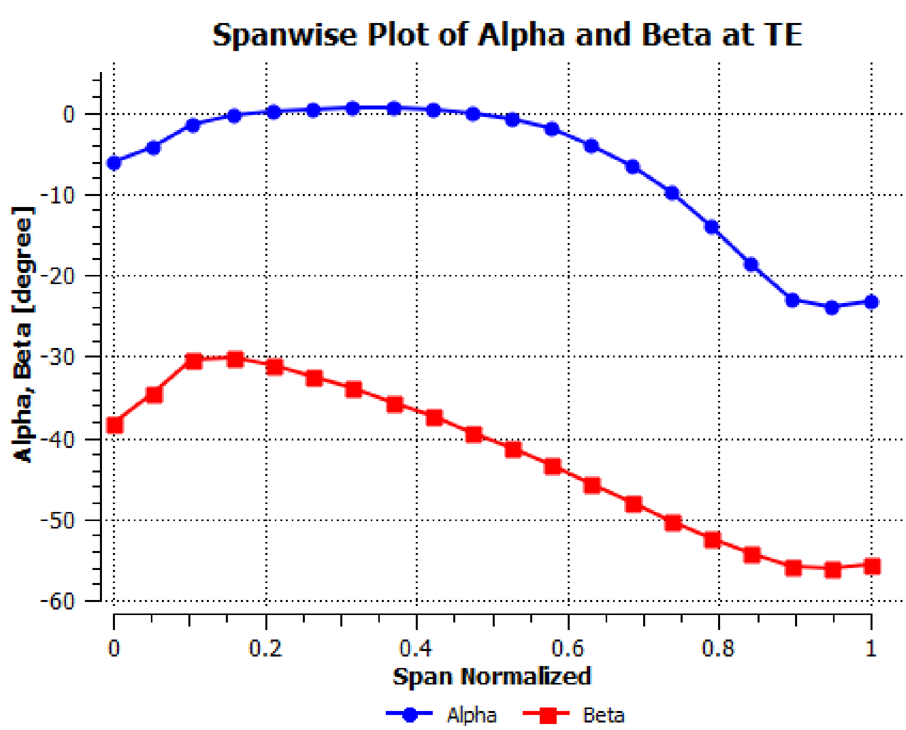
Observations

When the gas moves inward, the blade rotational velocity decreases with decreasing radius. So the gas velocity also decreases further. This results in gas to exit the impeller with low tangential velocity. Therefore the angular momentum of gas reduces. Due to this reason Thermodynamic properties drops from inlet to outlet.

Span wise plot of Alpha & Beta at leading edge



Span wise plot Alpha and Beta at trailing edge



Chapter 6

Conclusion & Future Work

6.1 Conclusion

Designing, meshing and Simulation of Turbo expander model has been done in Bladegen, Turbogrid and CFX respectively. After simulation, the results obtained were validated with the experimental results and various grpahs were plotted for various thermodynamic properties like Pressure, Temperature, Mach no, Velocity and density. According to graphs we observed that thermodynamic properties except Entropy, decreases along stream wise Inlet to Outlet. As the turbine blade velocity decreases according to decreasing radius the exit velocity of the flow too decreases. This causes in decreasing thermodynamic properties.

6.2 Future work

Computational Fluid flow analysis of rest parts like diffuser, nozzle etc may be done in future.

Different types of Turbo expander can be computationally analyzed in future works.

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